



Power generation from a new air-based Marnoch heat engine

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ABSTRACT

This paper examines the performance of a new Marnoch heat engine, which uses dry air and a pneumatic piston assembly to convert thermal energy to electricity. The system has unique capabilities of operating over temperature differentials less than 100 K. Unlike a common Stirling engine, the heat exchangers and piston assembly are not co-located, which is beneficial for positioning of heat exchangers in various configurations. This paper presents an operational laboratory-scale, proof-of-concept Marnoch heat engine (MHE), including its performance and power generation capabilities. It also presents a thermodynamic analysis of the system. Based on the MHE results, component modifications are made to improve its performance. The configuration has an efficiency of about thirty percent of a Carnot heat engine operating in the temperature range between 272 K and 372 K. Experimental data is acquired to provide verification of the predictive model, as well as demonstration of the MHE's capabilities for efficient generation of electricity from waste heat sources.

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1. Introduction

Rapid population growth in the world, at about 2% per annum, will create a large growing need for energy [1,2]. World energy demand in the past decade has increased by 20% in comparison to the 1990s [2]. Total daily world oil demand is predicted to increase from 77 million barrels in 2003 to about 119 million barrels in 2025, a 55% increase [3]. In 2004, a research study by the Pembina Institute in Canada [4] showed that electricity consumption during peak hours by the year 2020 could be reduced by 30% below 2004 levels, through a series of energy efficiency and demand reduction policies. The International Energy Agency (IEA) reported that more than 60% of the total worldwide energy demand is being supplied by fossil fuels [5]. In the U.S., the fossil fuel dependence was about 85% in 2007 [6]. Utilizing systems that work on more efficient thermodynamic cycles reduces the amount of pollution emitted to the environment [7]. Waste heat recovery from thermal power plants, industrial facilities, furnaces, and so forth, represents an important element of worldwide efforts to reduce greenhouse gas emissions and fossil fuel dependence [8]. Various types of heat engines are capable of recovering heat from lost energy. This paper focuses on heat recovery from a new type of heat engine, called a Marnoch heat engine (MHE).

Most gas power cycles were originally developed in the nineteenth century. Carnot and Stirling cycles are two well known power cycles [9]. In a Carnot cycle, the system undergoes a sequence of four reversible processes. This includes two adiabatic processes and two isothermal processes. The Stirling cycle represents the processes in a heat engine with regenerators (see Fig. 1). In this cycle, compressions and expansions are isothermal. Heat absorption and rejection in a Stirling cycle occur through constant volume processes [10]. In comparison with ICEs (internal combustion engines), heat engines have the advantages of operating on any heat source, not solely combustible fuels, simpler mechanisms, among others.

In Stirling engines, expansion and compression takes place in a cylinder with a power piston. A displacer transfers the working fluid through the heater, cooler, and regenerator. Fig. 2a shows the Alpha configuration of a Stirling engine. This assembly has no displacer, and the two pistons are working together so that volume stays constant. Cold and hot pistons move uniformly in the same direction and they provide constant volume heating/cooling of the working fluid. When the majority of gas is in one cylinder, the other piston expands and compresses the working fluid. Expansion occurs on the hot side, and compression on the cold side [11,12]. In the Beta configuration (Fig. 2b), a displacer and power piston are installed within the same cylinder [13]. The working fluid between the heat source, sink, and regenerator is moved by the displacer. The Gamma configuration (Fig. 2c) has two separated cylinders: one for the displacer and the other for the power piston [14]. In these configurations, the Stirling engine suffers from several disadvantages, including low specific power (large size with

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Nomenclature			
A	area [m^2]	Cs	cold source
AD	angular distance [o]	d	destroyed
B	bias limit	e	electric
C_{ave}	constant	f	final
Ex	exergy [J]	fa	from air
$\dot{Ex}$	exergy rate [W]	fpc	friction between sealing and the cylinder body
h	specific enthalpy [J/mol]	frp	friction between rack and pinion
H	enthalpy [J]	fw	from water
J_m	rotational mass moment of inertia [kgm^2]	gen	generated
k	radius of gyration [m]	Hs	heat source
KE	kinetic energy [J]	i	point number
L	length [m]	in	inlet
$\dot{Q}$	heat flow rate [W]	int	initial
M	mean value	loss	loss
m	mass [kg]	out	outlet
$\dot{m}$	mass flow rate [kg/s]	p	piston
N	number of experiments	Q	heat
P	precision limit	th	thermal
R	universal gas constant (J/mol·K)	to	total
S	entropy [J/kg K]	ta	to air
T	temperature [K]	tw	to water
t	time (s)	w	water
V	volume [m^3]		
U	uncertainty		
W	work output [J]		
X	piston length [m]		
Subscripts		Greek letters	
0	dead state (ambient condition)	Γ	exergetic temperature factor
1, 2, ...	state 1, 2, ...	τ	torque (N.m)
II	second law	η	efficiency
a	air	Δ	change
apl	all piston losses	σ	standard deviation
ave	average		
		Acronyms	
		NSP	number of strokes per minute
		Rt	rotational time [s]
		RPM	revolutions per minute
		SRDR	shaft rotation angle per revolution [°]

relative small power output), requires large temperature differentials to operate efficiently, and so on.

Various other types of low-temperature differential heat engines (LTDs) have been developed previously to generate mechanical power from low-temperature differentials. Abdullah et al. [15] developed a solar-based LTD, which generated power of about 100 W [15]. Kongtragool et al. [16] reported the characteristics of different LTD engines. It has been shown that the torque output of LTDs has been generally insufficient to drive industrial electric generators with high torque [17–19]. This paper presents a new type of heat engine, called a Marnoch heat engine (MHE), which has promising potential to overcome some limitations of past LTD technologies. It is capable of producing electricity from low grade heat sources and temperature differentials, below 100 K. Operation over such small temperature differences allows for a highly flexible system. This paper will acquire new experimental data and develop predictive models of the MHE performance, in order to demonstrate its promising performance as a new type of LTD for waste heat recovery.

2. Description of Marnoch heat engine

The Marnoch heat engine (MHE) operates similarly to other heat engines, by converting thermal energy to mechanical work, followed by electricity generation. The MHE consists of four cylindrical heat exchangers, each fitted with helical tube coil heat

exchangers. Each heat exchanger is connected to both a hot source and a cold sink, via tubes and valves that allow a controlled mass flow at any step of the process. Initially, all shells are kept at a constant pressure and mass. A half of the Marnoch cycle is driven by heat supplied to the exchangers, which drive pistons external to the exchangers. The other half of the cycle connects to a colder exchanger, facilitating thermal expansion of the gas in the hot exchanger and thermal contraction in the cold exchanger, via a piston assembly. The heated gas flows through the piston assembly of the heat engine and produces shaft work.

The configuration of the MHE with two pairs of heat exchangers is shown in Fig. 3. This figure shows the current assembly of the MHE. Fig. 4 indicates how the heat exchangers are connected internally and externally to the heat source and sink. When operation starts, hot water moves from the heat source through heat exchanger 1 and heats the gas (air) inside the shell. The air pressure inside the shell increases due to the temperature increase in the vessel. The coolant flows from the heat sink through to tubes of heat exchanger 2. As the result, it reduces the air pressure in the heat sink.

In the next step, when the air temperature inside the shells reaches a desired level, the valves are opened. Consequently, the air flows from the high pressure to low-pressure shell. Pipes are connected for a piston-cylinder assembly in series. Therefore, when air flows from the heated heat exchanger to the cooled heat exchanger, it moves the piston. When the piston reaches its end position, the

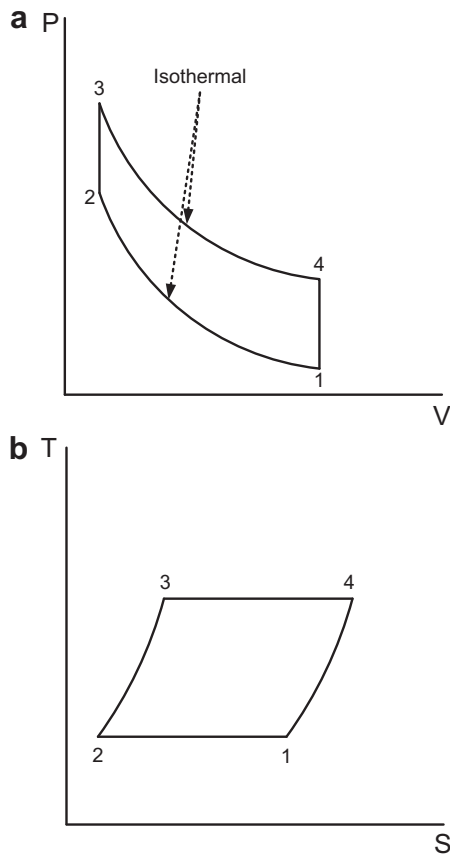


Fig. 1. Stirling cycle (a) P–V diagram; (b) T–s diagram.

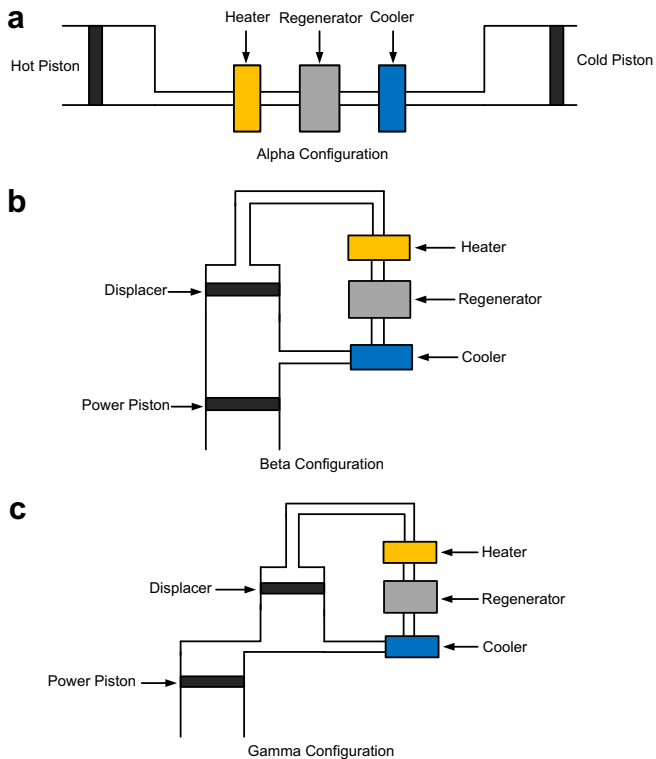


Fig. 2. Different configurations of Stirling engines.

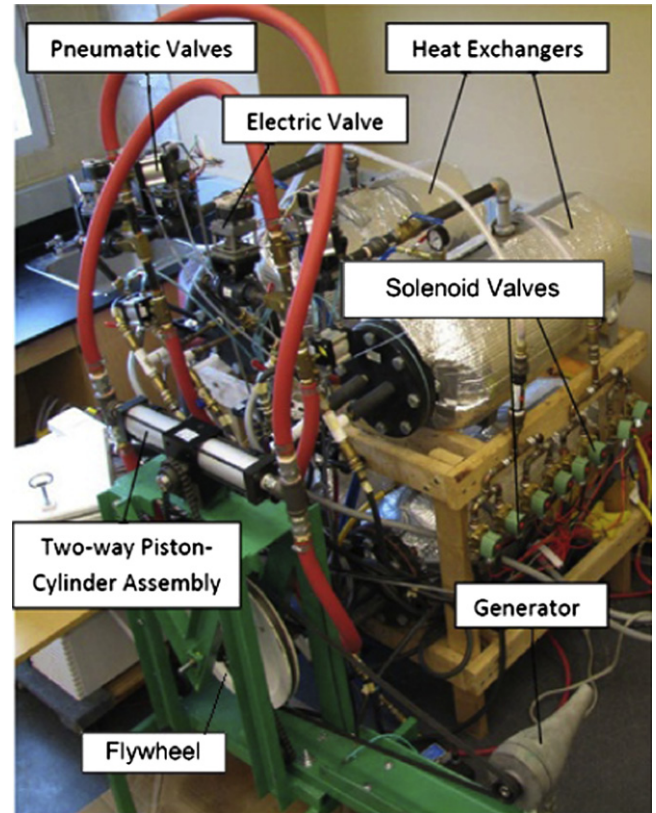


Fig. 3. Experimental apparatus of the Marnoch heat engine.

pneumatic valves redirect high pressure air to the other side of the piston to move it back to its initial position (see Fig. 4). The air moves from the high pressure shell to the heat sink, and it is stored in the cold shell.

By transferring air from the high pressure side and storing it in the cold heat exchanger, the pressure of the cold side increases

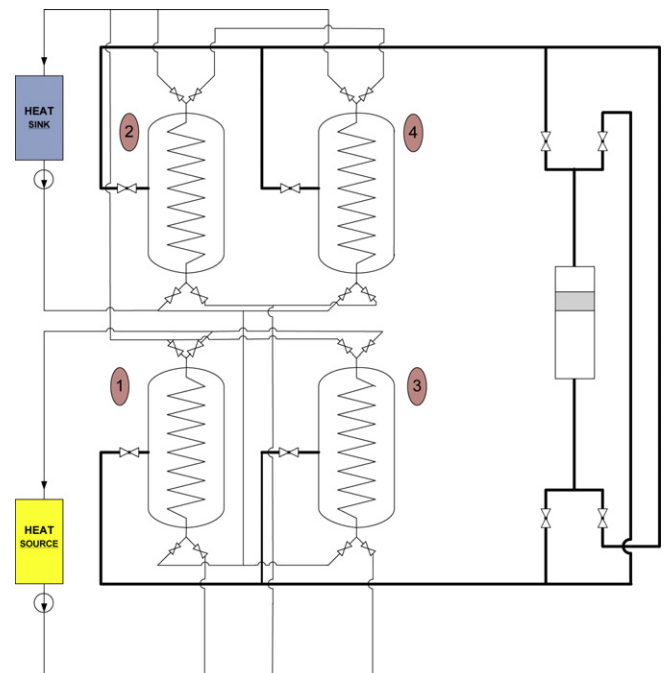


Fig. 4. Schematic of the MHE prototype with two pairs of heat exchangers.

gradually, and a larger mass of air accumulates on the cold side. After a certain number of strokes, based on the size of the exchangers and pre-charged pressure, the pressures of cold and hot sides become nearly equal. The air pressure is no longer sufficient to move the piston, after which a PLC (programmable logic controller) disconnects the first pair of exchangers and connects exchangers 3 and 4, for the remainder of the cycle.

In the final operation step, the roles of heat exchangers 1 and 2 change. During this period, the engine is using heat exchangers 3 and 4 to generate the pressure difference, for the engine to operate continuously. Throughout the second mechanical cycle of heat exchangers 1 and 2, heat exchanger 1 is used as the heat sink, and heat exchanger 2 is used as the heat source. Solenoid valves are changing the direction of water from the heat source and sink, such that hot water flows to heat exchanger 2 and cold water flows to heat exchanger 1 during the cycle. When the pressure inside heat exchangers 3 and 4 is nearly equalized, the first pair is reused to generate the temperature difference, and the last step is repeated for the second pair.

Thermodynamic and mechanical considerations show that there are major differences between the Stirling engine and MHE. In Stirling engines, a part of the mechanical work output from the flywheel is used in the compression process after each stroke. However, in the MHE, all of the energy output of the piston expansion is transferred to the flywheel. The reduced energy output due to the back pressure on the piston is equal to the compression energy to complete the thermodynamic cycle. Also, in MHEs, piston-cylinder assembly and heat exchangers can be located away from each other, whereas in the Stirling engine, the heat exchangers and piston assembly are attached. This feature of the MHE results in its ability to use a variety of sources to generate pressure differences from temperature differentials. Thirdly, unlike Stirling engines, after each stroke in MHEs, the enclosed area of a P–V diagram changes. This area is reduced, because after each stroke, the expansion pressure reduces and the compression pressure increases. Fourthly, in the mechanical configuration of the MHE, no regenerator is required. These differences will lead to noteworthy advantages of the MHE over a Stirling engine.

Fig. 7 demonstrates the sequential processes of the MHE cycle. Process 1–2: dry air inside the shell absorbs heat from the liquid flowing through the tube bundles. The shell is a rigid body, thus, the air volume stays constant and the gas pressure increases. Process 2–3: in this process, the valve that connects the high pressure shell to the low-pressure shell, via the piston assembly, is opened. Air expands and the piston moves forward. The expansion is almost isothermal, because the hot liquid flows through the tube bundle to keep the gas temperature constant during the expansion. Process 3–4: dry air accumulates in the low-pressure shell and it is cooled with the tube bundles that transfer cold water to the heat exchanger. The volume of the gas inside the shell stays constant and the pressure decreases due to the gas temperature change. Process 4–1: dry air is compressed with the piston and simultaneously cooled down with the liquid that flows through the tube bundles. This compression process is isothermal.

3. Experimental apparatus of the Marnoch heat engine

A proof-of-principle prototype was constructed, using a pinion rotary actuator assembly. The body of the cylinder is made of aluminum, which is hard-coat anodized and permanently sealed for wear resistance and endurance. The sealing in the system is critical to avoid leakages from the piston assembly to the surroundings, and from one side of the piston to the other side. Also, the friction loss between the piston and cylinder must be minimized. In the current experimental prototype, the maximum

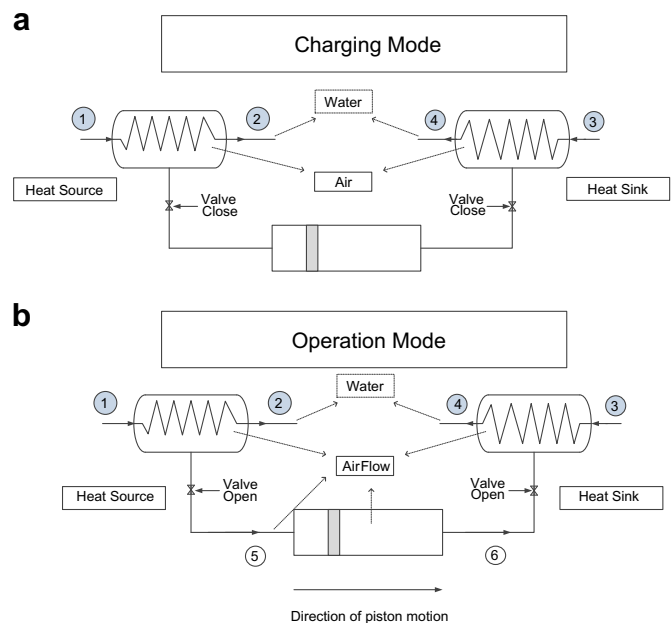


Fig. 5. Schematic of the MHE with one pair of exchangers for the thermodynamic analyses; section (a) is for the charging period and section (b) is for the discharging (operation) period.

operating pressure is 1723.7 kPa, with a breakaway pressure of 34.5 kPa.

Rotation of the shaft by each complete stroke is 270° . The displacement per degree of rotation in the cylinder is 1.23 cm^3 . The piston in this prototype has a diameter of 63.30 mm. Therefore, in

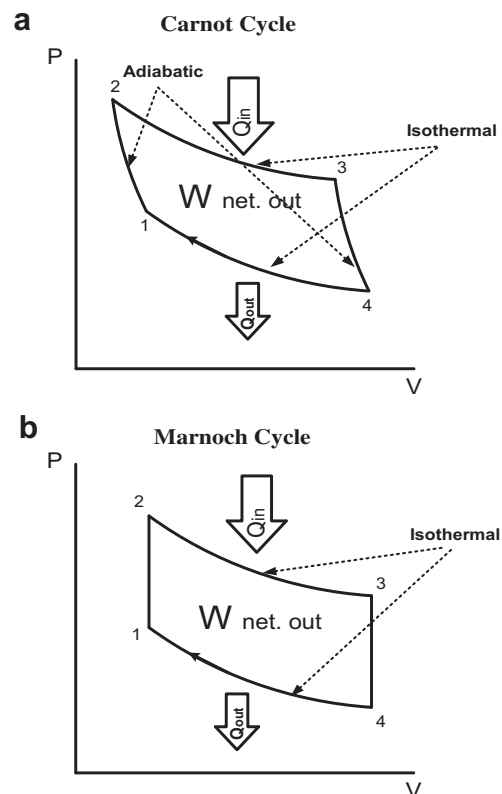


Fig. 6. Comparison of Carnot and Marnoch cycles.

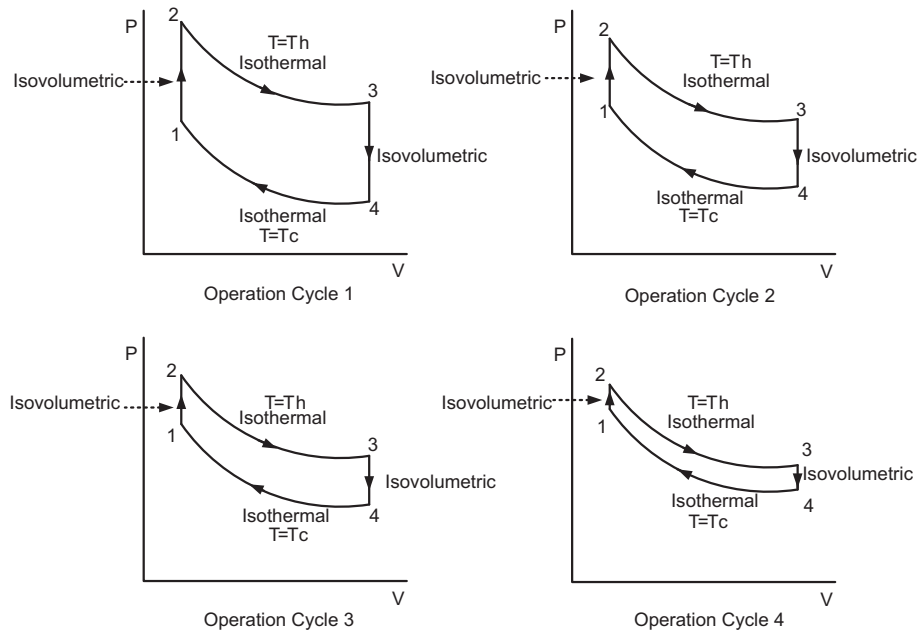


Fig. 7. Four power cycles of the MHE for five sequential piston revolutions.

each stroke, 331.83 cm³ of the working fluid is transferred to the low-pressure shell. Seals in the piston are designed to continually operate between temperatures ranging from 255 K to 355 K [20].

The torque outputs of the piston-cylinder, in four pressure differences, are shown in Table 1 for two different diameters. These values refer to a piston-cylinder with a diameter of 63.30 mm, in the current MHE, and they will be compared with the torque output from a piston assembly with a diameter of 81.21 mm. In Table 1, the higher torque output of the system requires higher pressure differences. For piston sizing of the MHE, if the volume of the cylinder is large, consequently the amount of working fluid transferred from one exchanger to the other is high, per stroke. Hence, if the volume of the heat exchangers does not match the piston size after a few strokes, the engine operation will stop and another pair of exchangers must be charged for operational use. The charging mode is time consuming, and delays in piston operation happen when exchangers are switched. When a larger piston-cylinder is used, less time is required for changing the direction of the piston through the valves. Thus, the power output from the piston assembly will be higher. Moreover, the torque output is more than the output of smaller pistons. When smaller pistons are used, the direction must be changed more frequently. Changing the direction of valves is time consuming, so a specified amount of electricity is produced over a longer duration. Therefore, the power output of the piston is lower.

The inaccuracies related to measurement devices are bias errors. The measurement accuracy of the pressure gauges and power meter is about $\pm 1\%$. Random variations are errors that occur due to uncontrolled parameters when making repeated measurements. They are related to the precision of the experimental readings and

determined by finding the deviation of readings. The root-sum-square method was applied to find the measurement uncertainties. Using the method of Kline and McClintock [21], the measurement uncertainty (U) for the experimental data is calculated by the following equation:

$$U = \sqrt{B^2 + P^2} \quad (1)$$

where B is the bias limit, and P is the precision limit for the experimental readings. Bias is the error associated with the inaccuracy of the measuring equipment. The precision can be found based on the following equation [22]:

$$P = \frac{\sigma}{\sqrt{N}} \quad (2)$$

where σ and N are the standard deviation and the number of experiments, respectively. The measurement uncertainties will be reported in the next section, alongside experimental results that are presented in the figures. The estimated error in the experimental data is approximately $\pm 4.6\%$. This will place the experiment within a 95% confidence interval (see Table 2).

4. Thermodynamic analysis

Both energy and exergy analyses are performed to evaluate the performance of the MHE [23]. The schematic of the system is shown in Fig. 5, which represents the first operation cycle in two different

Table 1
Torque output of the piston [20].

Pressure difference (kPa)	344.74	517.11	689.475	1723.69
Torque output (N.m)	24.29	36.38	48.58	121.35
(Piston diameter: 63.30 mm)				
Torque output (N.m)	64.40	96.04	128.92	322.23
(Piston diameter: 81.21 mm)				

Table 2
Estimated uncertainty of the measured power output from the generator and actuator.

	Precision error (W)	Bias error (W)	Uncertainty (W)	Uncertainty (%)
Total power output from generator	48.5	19	52.1	4.3
Total power output from actuator	114.9	47	124.2	4.0
Peak power output from generator	3.6	1.0	3.8	4.6

stages (charging and discharging). In Fig. 5a, both air valves are closed and initially both shells have the same pressure. When water flows through the heat exchangers, heat exchanger 1 is heated to raise the pressure, and at the same time, heat exchanger 2 is cooled and its pressure decreases. In the second step (Fig. 5b), the air temperatures inside both heat exchangers are close to the temperatures of the water streams. Air valves are opened and the set of heat exchangers and piston start their operation. Operation of this set continues until the air pressures inside both heat exchangers equalize. During charging and discharging periods, water flows through the heat exchangers continuously, and it does not shut off until the last stroke. This occurs to achieve near isothermal expansion and compression strokes of the piston.

4.1. Mass and energy analyses

Heat exchanger 1 is a shell and tube heat exchanger with both fluids unmixed. A heating fluid (hot water) flows through the tubes and exchanges heat with the working fluid (air) in the shell side. The overall mass balance can be written as

$$\Delta \dot{m} = \sum \dot{m}_{in} - \sum \dot{m}_{out} \quad (3)$$

Eq. (3) is written separately for water as

$$\dot{m}_{w1} = \dot{m}_{w2} \quad (4)$$

For air, the system is not at steady state, and the conservation of mass requires

$$\frac{m_{Air,out}}{dt} + \sum \dot{m}_5 = 0 \quad (5)$$

From Eq. (5) the total amount of air mass that leaves the hot heat exchanger can be calculated (see Fig. 5b). Integrating over time gives the change of mass in heat exchanger 1 during the overall process:

$$\int_0^t \left(\frac{m_{Air,out}}{dt} \right) dt = m_{initial} - m_{final} \quad (6)$$

In Eq. (4), $m_{initial}$ denotes the initial air mass in shell 1, and m_{final} denotes the air mass that remains in shell 1 after one cycle of operation. For the whole operation, over a period of time (t), the continuity equation for the transient process is written as

$$(m_{initial} - m_{final}) = - \sum \dot{m}_5 (\Delta t) \quad (7)$$

In the first cycle of operation, as shown in Fig. 5, heat exchangers 1 and 2 are connected to the hot and cold reservoir, respectively. The general energy balance is given by:

$$\Delta \dot{E} = \sum \dot{E}_{in} - \sum \dot{E}_{out} \quad (8)$$

where $\Delta \dot{E}$ is the total energy change in the shell and tube heat exchanger. This includes the energy change in both liquid and gas states. Also, $\sum \dot{E}_{in}$ is the summation of the total energy input to the heat exchanger, and $\sum \dot{E}_{out}$ is the energy that leaves the heat exchanger control volume. This equation is used for heat exchangers 1 and 2 for both modes: charging and discharging. In the first step for heat exchanger 1, heat is transferred into the system by the hot water stream. A portion of its energy is transferred to the gas inside the heat exchanger. A fraction of $\sum \dot{E}_{in}$ goes back to the surroundings by the exit water stream, and the remainder is lost to the surroundings. Thus, the energy balance for the water stream becomes

$$\dot{m}_{w1} h_{w1} = \dot{m}_{w2} h_{w2} + \dot{Q}_{ta} + \dot{Q}_{loss} \quad (9)$$

where the values of $\dot{m}_{w1}$ and $\dot{m}_{w2}$ are obtained from Eq. (4), and h_{w1} and h_{w2} denote the enthalpy of water that enters and exits the heat exchanger, respectively. The term $\dot{m}_{w2} h_{w2}$ (energy flow to the ambient via the outlet water stream) changes over time and its value depends upon $\dot{Q}_{ta}$ (heat transfer rate from the water to the working fluid). This rate decreases as the temperature of the gas approaches the water temperature. Also $\dot{Q}_{loss}$ is the amount of heat rejected to the ambient from the hot stream tubes, and it also varies with time because the fluid temperature changes with time. The heat transfer rate ($\dot{Q}_{ta}$) from the water stream to the air is expressed as

$$\int_0^t \dot{Q}_{ta} (dt) = mc_v \Delta T \quad (10)$$

When Eq. (10) is used for the charging period, the air mass (m) is constant inside the shell, and its value is $m_{initial}$ in Eq. (7).

When the piston operation starts, in Fig. 6, it is assumed that $T_1 = T_2 = T_5$, where T_1 and T_2 are the water inlet and outlet temperatures respectively, and T_5 is the gas temperature inside the hot heat exchanger. The heat loss from the pipe, which connects shell 1 to the piston assembly, is neglected (well insulated system). Hence, air temperatures at point 5 (Fig. 5) and inside shell 1 are the same.

When air leaves the hot heat exchanger, the pressure of the shell decreases. The expansion process in this heat engine is kept as close as possible to isothermal to obtain maximum work output. Water flows through the heat exchangers during piston operation, so when the gas is expanded, it is being heated simultaneously. The energy balance equation inside exchanger 1 during the operation mode can be expressed as

$$\Delta \dot{E} = \dot{Q}_{ta} - \dot{m}_5 h_5 \quad (11)$$

where $\Delta \dot{E}$ is the rate of energy change in exchanger 1 due to heat and mass transfer.

4.2. Exergy analysis

To determine the performance of the system at different ambient conditions, an exergy analysis is also performed. The physical exergy of each point Ex_i can be calculated by

$$Ex_i = (H_i - H_0) - T_0(S_i - S_0) \quad (12)$$

where H_0 , T_0 and S_0 are enthalpy, temperature and entropy of air under ambient conditions, respectively. Also H_i and S_i are the enthalpy and entropy at each point within the system boundary. The thermal exergy content is given by:

$$\dot{Ex}_Q = \int_{int}^f \left(1 - \frac{T_0}{T} \right) dQ \quad (13)$$

where dQ is the incremental heat transfer, and the integral is taken from the initial state (int) to the final state (f). This thermal exergy is the minimum work required in bringing the system to its final state from the dead state. In the MHE prototype, under most conditions, heating the exchanger starts from the dead state.

The dimensionless quantity $(1 - \frac{T_0}{T})$ is called the “exergetic temperature factor”, and it is denoted as [24]:

$$\Gamma = \left(1 - \frac{T_0}{T} \right) \quad (14)$$

For heat transfer across a segment (r) of the control surface, for which shell 1 is charged initially, the thermal exergy associated with heat transfer is

$$\dot{Ex}_Q = \int_r q_r \cdot T dA_r \quad (15)$$

where q_r is the heat flow per unit area at a region on the control surface, and the temperature is denoted as T_r . The dimensions of the copper tubes inside the heat exchanger are used to find the surface area (see Table 3). If the temperature (T) of the control mass is constant, the thermal exergy transfer associated with heat transfer is

$$\dot{Ex}_Q = \left(1 - \frac{T_0}{T}\right) \dot{Q}_{ta} = T \dot{Q}_{ta} \quad (16)$$

where T_0 is the ambient temperature and T is the working fluid temperature during the charging mode.

4.3. Shaft work output of the piston assembly

In Fig. 6, the thermodynamic power cycle of the MHE is shown and compared with an ideal Carnot cycle. Unlike the Stirling cycle in Fig. 6b, the enclosed area of the MHE is reduced after each stroke during the operation of one exchanger pair. Fig. 7 is a demonstration of the area reduction of cyclic work output from the piston assembly. Pre-charge pressures and available temperature differences will affect the number of piston cycles, and number of power cycles per operation period. The compression and expansion processes in the MHE are assumed isothermal, and air is assumed to be an ideal gas. The ideal gas law ($PV = nRT$) will be assumed to find the air properties. In this equation, P is the absolute pressure of the air, n is number of moles of air and R is the universal gas constant which is $8.314 \text{ (J/mol} \cdot \text{K)}$.

The MHE has a closed cycle, so no gas enters or leaves the system during the operation cycle. Hence, the number of moles of air (n) is constant. Also, the expansions and compressions are nearly isothermal. Therefore, the value of nRT is nearly constant, which is shown by C in Eq. (17). Thus, the ideal gas equation implies:

$$P = \frac{C}{V} \quad (17)$$

The work output of the piston as the result of one expansion can be found as

$$W = \int_{V_{int}}^{V_f} P dV \quad (18)$$

where P is a function of the volume.

Substituting Eq. (17) into (18) yields

$$W = \int_{V_{int}}^{V_f} \frac{C}{V} dV = C \int_{V_{int}}^{V_f} \frac{1}{V} dV \quad (19)$$

$$W = C \ln\left(\frac{V_f}{V_{int}}\right) = C \ln\left(\frac{V_{int} + \Delta V}{V_{int}}\right) = C \ln\left(1 + \frac{\Delta V}{V_i}\right) \quad (20)$$

The piston surface is constant during operation, but the length changes. Using $\Delta V = A \Delta L$ in Eq. (20), it results in

$$W = C \ln\left(1 + \frac{A \Delta L}{V_i}\right) = C \ln(1 + C_{ave} \Delta L) \quad (21)$$

where C_{ave} is a constant that is defined as $C_{ave} = \frac{A}{V_i}$. Each piston assembly in the MHE has a specific value of C_{ave} .

Eq. (20) shows the work output of the piston after one cycle. Then some air mass is transferred to the other exchanger. When the work output of the next stroke of the piston is calculated by Eq. (20), the work performed by the compression of the piston is negative.

4.4. Analysis of transmission system

Table 2 shows the torque output from the current piston assembly at certain pressure differentials, based on the manufacturer's specifications [20]. To find the rotational velocity of the piston, the following equation is used:

$$RPM_p = \frac{(NSP)(SRDR)}{360} \quad (22)$$

where RPM_p is the piston shaft rotation per minute, NSP is the number of strokes per minute and $SRDR$ is the shaft rotation degrees per revolution. In the current piston assembly, the pinion rotates 270° after each revolution.

To find the power output for the piston, RPM_p is multiplied by the torque output from the piston assembly,

$$Power_p = \tau_p \cdot (RPM_p) \quad (23)$$

where τ_p is the torque output of the piston assembly. The value τ_p changes after each revolution, because the pressure differences after each stroke decrease. In order to estimate the power output from the piston assembly, the output kinetic energy from the piston to the flywheel is calculated. Determining the losses in chains and bearings, the energy output from the piston can then be estimated.

The thickness of the disc is negligible in comparison to its diameter. Thus, the kinetic energy from the flywheel is written as

$$KE = \frac{1}{2} J_m \cdot \omega^2 \quad (24)$$

where KE is the kinetic energy of the flywheel, J_m is the rotational mass moment of inertia, and ω is the angular velocity. As given below:

$$J_m = \frac{w}{g} \cdot \frac{k^2}{2} \quad (25)$$

where w is the weight load, g is the gravitational constant, and k is the radius of gyration. The value of ω to be substituted in Eq. (24) is calculated by

$$\omega = \frac{AD}{Rt} \cdot (0.035) \quad (26)$$

where AD is the angular distance and Rt shows the rotational time.

Table 3
Helical coil specifications.

Thickness of tube	0.64 mm
Inner diameter before deformation	10.92 mm
Outer diameter before deformation	12.2 mm
Number of coils	37
Length of heat exchanger coils	630 mm
Radius of bends	65 mm
Total length of tubes	17,350 mm
Total occupied volume by tubes	$2006.23 \times 10^{-6} \text{ mm}^3$
Percentage of occupied shell volume by tubes	2.57%

4.5. Efficiencies of the Marnoch heat engine

The thermal efficiency of the system is given by

$$\eta_{th} = \frac{W_{output}}{Q_{Hs}} \quad (27)$$

where W_{output} is the work output of the cycle, and η_{th} is the thermal efficiency of the system. The variable W_{output} can be calculated using Eq. (19). The value Q_{Hs} is the heat flow into the system from the heat source. It is defined as the total heat flow into the system from the hot water stream, which can be calculated by substituting Q_{ta} (amount of heat transferred to the air) into Eq. (27). Also, the overall efficiency of the system (η_{to}) can be expressed as

$$\eta_{to} = \frac{W_e}{Q_{Hs}} \quad (28)$$

where W_e is the electric work output from the engine.

The Carnot efficiency of the MHE is

$$\eta_{ideal} = 1 - \frac{T_{Cs}}{T_{Hs}} \quad (29)$$

where the values of T_{Cs} and T_{Hs} are the temperatures of the heat sink and heat source, respectively.

The second law provides an indication of the irreversibilities within the system. The cycle exergy efficiency of the MHE based on the second law is defined as

$$\eta_{II} = \frac{\eta_{th}}{\eta_{ideal}} \quad (30)$$

This can be calculated by substituting the cycle efficiency of the engine and Carnot efficiency into Eq. (30). The exergy efficiency provides an alternative interpretation of performance. When evaluating the energy efficiency, the same weight is assigned to energy whether it is electric, shaft work or low-temperature thermal energy. The exergy efficiency weighs the energy flows with respect to exergy [25]. Using the exergy efficiency, it can be better understood how internal and external irreversibilities affect the performance of the engine. The exergy efficiency (η_{Ex} ; same as second law efficiency) for the MHE is defined as

$$\eta_{Ex} = \frac{W_e}{Ex_{in}} \quad (31)$$

Ex_{in} is the exergy entering the system. To calculate Ex_{in} , the exergy of hot water can be obtained from Eq. (12) (point 1 in Fig. 5). Thus, the value of Ex_{in} is the exergy of the hot water stream that flows through the hot heat exchanger. Alternatively, Eq. (16) can be used to find Ex_{in} . Considering Eq. (16), the denominator is smaller than the denominator of Eq. (28). Hence, the exergy efficiency of the system is higher than the system energy efficiency. This can be understood by low grade energy (such as waste heat) in the MHE that is converted to higher grade energy in the form of electricity.

5. Results and discussion

In this section, experimental data is obtained for the Marnoch heat engine, and compared against predictions from the thermodynamic model in the previous section. Fig. 8 shows how the pressure measurements vary within one pair of heat exchangers (hot and cold) during the operating period. This figure shows that the engine starts operation when the air pressure inside the first shell was 882.53 kPa, and the pressure of the other shell was 103.42 kPa. Before starting the experiment, there was a 160 s delay. Within this period, as shown in Fig. 8, the pressure inside

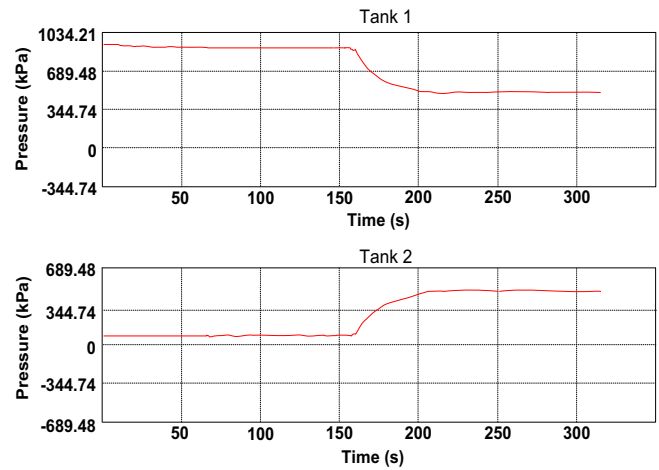


Fig. 8. Pressure variations in one pair of MHE heat exchangers (gauge pressure).

exchanger 1 dropped. The temperature of compressed air was initially higher than the ambient temperature, and the shell side of the heat exchanger was not perfectly insulated. Heat transfer from the air to the shell side, and from the shell side to the surroundings, occurred. In exchanger 2, the initial pressure inside the shell was close to the ambient pressure, so within the first 160 s, no considerable pressure change was observed in exchanger 2. The results in Fig. 8 refer to one operational cycle of the MHE and the measurement uncertainty (U) in the reported data is ± 10 kPa.

When the engine operation starts, due to heat transfer and air flow from exchanger 1 to exchanger 2, the pressure of exchanger 1 decreases, and the pressure of exchanger 2 increases, simultaneously. The curved sections in Fig. 8 show this period. When the piston cycle ends, the pressures inside the shells did not equalize. The remaining pressure difference was about 20.68 kPa in the experiment. This occurs because of friction losses in the piston assembly, so the remaining pressure difference could not move the piston. The pressure differences were determined from the experiments and predictions, to find the power output from the piston assembly and generator. The ideal gas law was used to predict the pressure differentials in the heat engine at various operating temperatures.

Before each test, shells were filled with compressed air at different pressures. During the experiments, hot and cold water passed through the heat exchangers, in order to heat and cool them. Fig. 9 shows the total measured power output from the electric generator during one operation cycle. To obtain values for each pressure differential, the engine was operated at least three times for each initial pressure differential. The results shown in Fig. 9 are the average readings of one set of experiments. When the engine works under high pressure differences, the trend line in Fig. 9 shows the electricity generated from the system increases.

Fig. 10 shows the total mechanical power output from the piston assembly at specified pressure differences. The mechanical power outputs are determined by measuring the number of piston revolutions, within a certain time period, and the piston specifications from the manufacturer. The number of piston revolutions was found by measuring the number of changes (open/close) of the pneumatic valve. Comparing Figs. 9 and 10, the amount of power loss within the transmission system of the MHE can be observed.

Fig. 11 shows the measured power output from the generator during one cycle of operation. The power is measured after each piston revolution [26]. The measured values for each curve in the graph represent one set of operating conditions of the heat engine. The parameter that has an effect on the measurement uncertainty

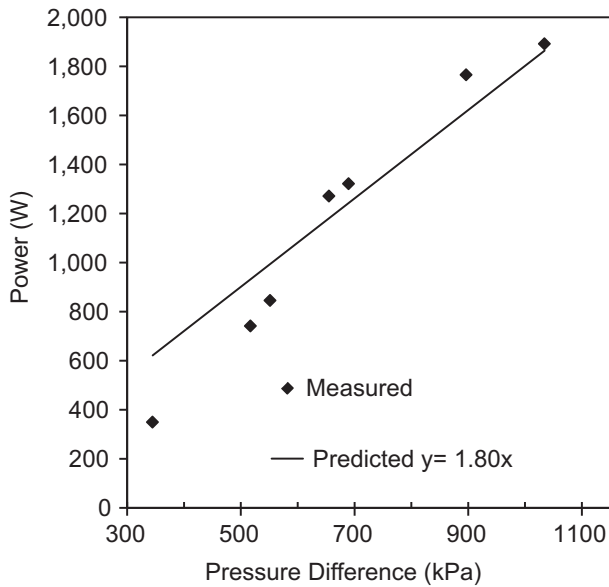


Fig. 9. Total power output from the generator at varying pressure differences.

is the instrumentation error. The measurement error of the instrument in the current setup is ± 0.95 (W). The minimum pressure differential for these tests was 344.74 kPa, and the maximum was 1034.21 kPa. The power output from the generator shows fluctuations during the operation cycle, due to losses occurring in the flywheel. During operation at certain times, the speed of the flywheel is higher than the piston speed. Consequently, the energy output of the piston is not added to the flywheel energy. When the flywheel speed decreases, the power of the piston is added to the system again.

In Fig. 12, the points represent the measured power output from the generator at two different pressure differentials. The continuous line shows the predicted power output from the generator at a 779.11 kPa initial pressure differential. The experimental pressure readings for this power prediction are presented in Fig. 8. To validate the predicted power output from the

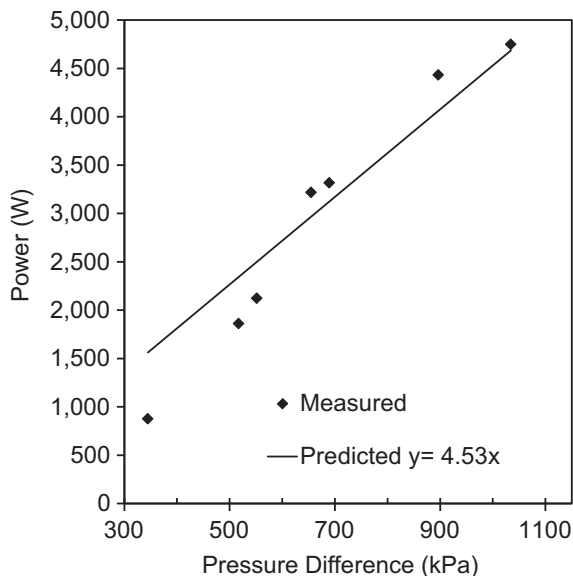


Fig. 10. Total power output from the piston assembly at varying pressure differentials.

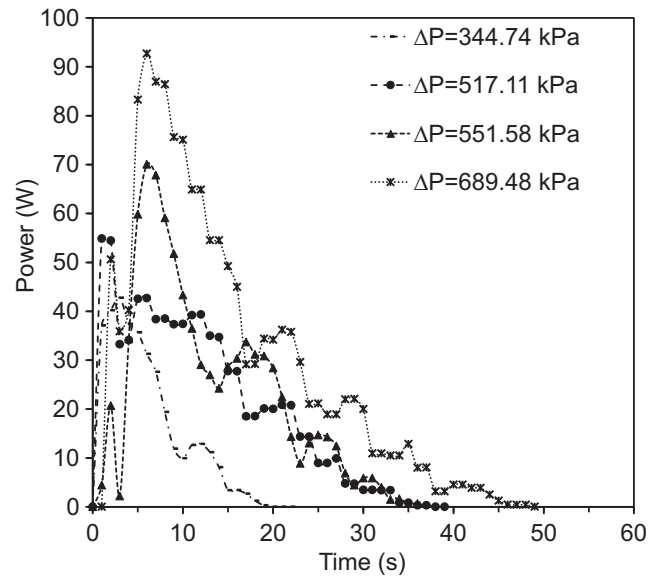


Fig. 11. Measured power output with one exchanger pair at varying pressure differentials.

generator, the power output measurements from the upper and lower pressure differentials were used. This line is located between the power output readings of the initial pressure difference of 689.48 and 1034.48 kPa.

The predicted power output starts from a maximum value and drops over time gradually. However, the power measurements start from almost zero, increases higher, drops drastically and then the power reaches its maximum. The reason for this trend is the initial energy required to rotate the flywheel. This trend is visible in most of the MHE measured data (see Fig. 12).

Fig. 13 shows a comparison of measured and predicted peak power outputs from the generator for varying pressure differentials. Seven observed peak power outputs at certain initial pressures are shown in this graph. The continuous solid line shows the predicted peak power outputs from the generator. The closest

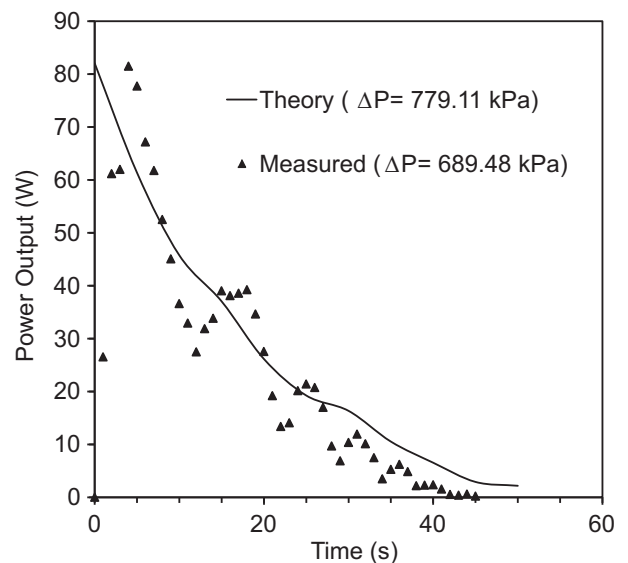


Fig. 12. Comparison of measured and predicted power outputs from the generator for varying pressure differentials during one operation cycle.

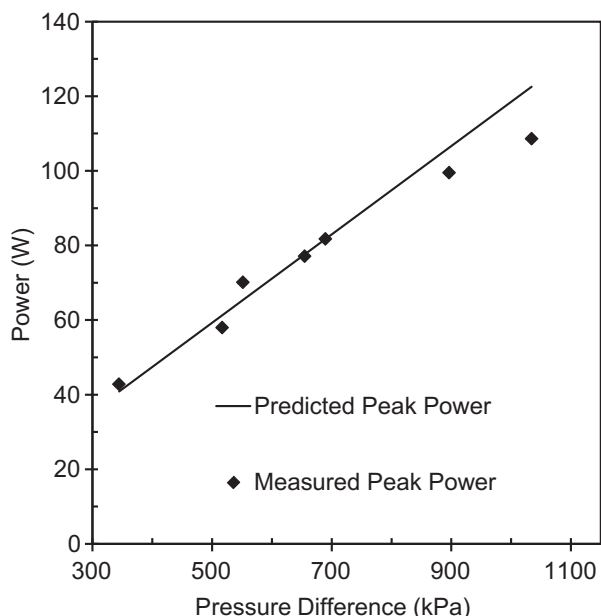


Fig. 13. Comparison of measured and predicted peak power outputs from the generator for varying pressure differentials.

match between the model and measured data occurs between the initial pressures of 600 and 700 kPa. At pressures around 1000 kPa, the model predicts higher power outputs from the system than the measured data. This occurs because the transmission efficiency was measured at 689.48 kPa. At higher pressure differentials, the rotary speed of the flywheel increases. Thus, more energy from the piston revolutions is not captured. At high pressure differentials, the transmission efficiency is less than 40%.

Fig. 14 shows the Carnot efficiency of the system. The temperature ranges for the heat source and heat sink in this study are between 272 K and 373 K. In this range, water can add or remove heat from the system. In addition, in many areas of industry, waste heat in this temperature range is freely available. In Fig. 14, each line

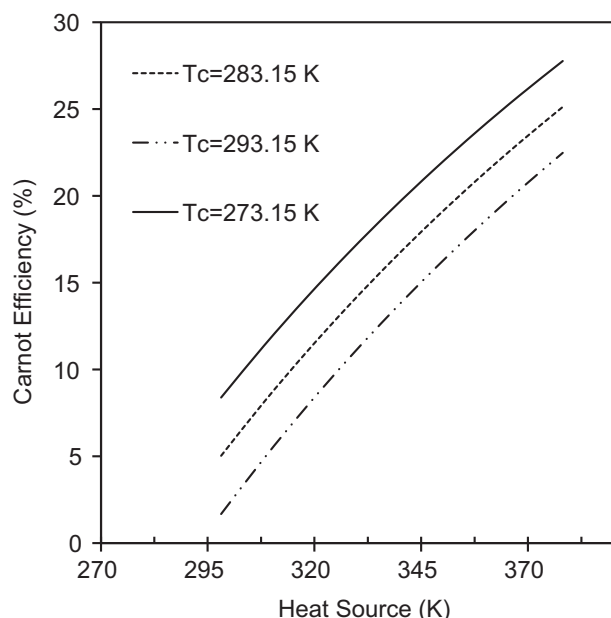


Fig. 14. Variation of Carnot efficiency at different source temperatures.

demonstrates the change of Carnot efficiency by changing the heat source temperature at a constant heat sink temperature. The main trend of the curves shows an increase in Carnot efficiency by elevating the source temperature. For instance, at a temperature of 293 K and 298 K, for heat sink and source temperatures, the Carnot efficiency of a heat engine is 1.7%. When the temperature of the heat source is 353 K, the efficiency reaches 22.5%. The figure demonstrates that having the same temperature differences in lower temperature ranges gives a higher Carnot efficiency of the heat engine. When the heat sink temperature is 273 K, the efficiency of the heat engine is higher than 283 K and 293 K. This trend is valid for temperature differentials below zero as well. Also, in Fig. 14, the maximum possible efficiency achievable from this system in temperature ranges of 273 K–373 K is 27.8%. Consequently, when water is used to add and remove heat from the system, the thermal efficiency of the system cannot exceed 27.8%.

In the Marnoch heat engine, exergy losses occur in the system due to heat losses and mechanical component deficiencies. The electric power from the MHE is the exergy recovered by this heat engine. About 60% of the mechanical power output from the piston is lost due to friction in chains, belt, ball bearings and flywheel [26]. The existing mechanical loss in the MHE reduces the exergy output (recovered). In the current experimental apparatus, the heat loss from the heat exchangers is a significant portion of the total heat input to the heat exchangers [27]. When the ambient temperature is 298 K and the heat is supplied to the engine at a temperature of 360 K, the dimensionless exergetic temperature factor is 0.17. It is assumed that the heat engine operates at an atmospheric pressure of 101.3 kPa. This indicates the exergy flow rate associated with the heat loss, Ex_{loss} , at atmospheric conditions is about 183.3 W.

6. Conclusions

This paper has presented, analyzed and performed experimental studies to demonstrate the promising performance of a new type of low-temperature heat engine, called a Marnoch heat engine. The heat engine is an energy conversion device that has the capability to recover low grade heat to electricity, using smaller temperature differentials than other existing methods. The MHE has strong potential to be used in many applications, such as waste heat recovery from thermal power plants, industrial facilities, furnaces and others. This paper presented new data for the thermal efficiency and power output of the MHE, as well as a predictive model that was validated against the experimental measurements. The studies indicate that higher pressure units will result in larger power output, higher efficiency and better economic viability of the system.

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References

- [1] Dincer I. Renewable energy and sustainable development: a crucial review. *Renewable and Sustainable Energy Reviews* 2000;4:157–75.
- [2] Dincer I, Rosen MA. Thermodynamic Aspects of renewables and sustainable development. *Renewable and Sustainable Energy Reviews* 2005;9:169–89.
- [3] Rasthal JE, Drennen TE. *Pathways to a Hydrogen Future*. 1st ed. UK: Elsevier; 2007.
- [4] Etcheverry J. Comments regarding the ontario power authority's (OPA). Supply Mix Advice Report. Toronto: David Suzuki Foundation; February 2006.
- [5] Anon. World energy council. Technical Report", *Global Energy Perspectives to 2050 and Beyond*, London; 1995.
- [6] Randolph J, Masters GM. *Energy for sustainability*. 1st ed. , Washington: Island Press; 2008.

- [7] Sun ZG. Experimental investigation of integrated refrigeration system (IRS) with gas engine, compression chiller and absorption chiller. *Energy* 2008;33:431–6.
- [8] Ahmadi P, Dincer I. Exergoenvironmental analysis and optimization of a cogeneration plant system using Multimodal Genetic Algorithm (MGA). *Energy* 2010;35:5161–72.
- [9] Rolle KC. Thermodynamics and heat power. 6th ed. Virginia: Pearson/ Prentice Hall; 2005.
- [10] Moran MJ, Shapiro HN. Fundamentals of Engineering Thermodynamics. 6th ed. New York: Wiley; 2007.
- [11] Kongtragool B, Wongwises S. A Review of solar-powered Stirling engines and low temperature differential Stirling engines. *Renewable and Sustainable Energy Reviews* 2003;7:131–54.
- [12] Berchowitz DM, Shonder J. Estimated size and performance of a Natural gas Fired Duplex Stirling for Domestic Refrigeration applications. Technical report. Oak Ridge, Tennessee, U. S. A: Oak Ridge National Labs; October 2008.
- [13] Karabulut H, Aksoy F, Ozturk E. Thermodynamic analysis of a Beta type Stirling engine with a displacer driving mechanism by means of a Lever. *Renewable Energy*; 2009:202–8.
- [14] Parlak N, Wager A, Elsener M, Soyhan HS. Thermodynamic analysis of a Gamma type Stirling engine in Non-ideal adiabatic conditions. *Renewable Energy*; 2009:266–73.
- [15] Abdullah S, Yousif BF, Sopian K. Design consideration of low temperature differential Double-acting Stirling engine for solar Application. *Renewable Energy*; 2005:1923–41.
- [16] Kongtragool B, Wongwises S. A four power-piston low-temperature differential Stirling engine using Simulated solar energy as heat source. *Solar Energy* 2008;82:493–500.
- [17] Senft JR. Mechanical efficiency of heat engines. 1st ed. New York: Cambridge University Press; 2007.
- [18] Ramesh CV. Valved heat engine working on Modified Atkinson cycle. *ASME Journal of Energy Resources Technology* 2010;132.
- [19] Zao Y, Lin B, Chen J. Optimum Criteria on the important parameters of an Irreversible Otto heat engine with the temperature-dependent heat Capacities of the working fluid. *ASME Journal of Energy Resources Technology* 2007;129: 348–54.
- [20] Pneumatic Automation Products, manufacturer Catalogue. USA: Parker Hannifan Corporation; 2001.
- [21] Kline SJ, McClintock FA. Describing uncertainties in single-sample experiments. *Mechanical Engineering* 1953;75.
- [22] Abernethy RB, Benedict RP, Dowdell RB. ASME measurement uncertainty. American Society of Mechanical Engineers; 1983.
- [23] Ozgener L, Hepbasli A, Dincer I. Thermo-Mechanical exergy analysis of Balcova Geothermal District heating system in Izmir, Turkey. *ASME Journal of Energy Resources Technology* 2004;126:293–301.
- [24] Dincer I, Rosen MA. Exergy: energy, environment and Sustainable Development. 1st ed. Elsevier; 2007.
- [25] Uche J, Serra L, Valero L. Exergy costs and inefficiency diagnosis of a dual-purpose power and desalination plant. *ASME Journal of Energy Resources Technology* 2006;128:186–93.
- [26] Self S. Power testing of the Prototypical Marnoch thermal energy conversion device. Technical Report. Oshawa, Canada: Faculty of Engineering and applied Science, University of Ontario Institute of Technology; April 2008.
- [27] Saneipour P, Naterer GF, Dincer I. Heat recovery from a cement plant with a Marnoch heat engine. *Applied Thermal Engineering* 2011;31:1734–43.